Jurnal Mekanikal
December 2009, No.29, 100-111

EFFECT OF ACCELEROMETER MASS ON THIN PLATE VIBRATION

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ABSTRACT

Generally accelerometer mass effect is ignored in experimental studies based on the assumption that the accelerometer mass is negligible compared to the mass of the test structure. However when lightweight structure is tested, this effect is significant. The purpose of this study is to investigate the effect of accelerometer mass on thin plate vibration. The natural frequency and its corresponding mode shape were the parameters of interest. The thin plate clamped free boundary condition was investigated by using experimental modal analysis and finite element analysis. The accelerometer mass was added and mounted at three different locations. Results show the accelerometer mass has significant effect on some of the modes of the structure while other modes remain unchanged. The accelerometer mounted at a point of peak deflection of the plate showed large changes of natural frequencies and its corresponding mode shapes. There are no significant changes of natural frequency and mode shapes for accelerometer attached at nodal line of the particular mode. It is concluded that, the effect of accelerometer mass depends on the location of the accelerometer, vibration mode and magnitude of accelerometer mass.

Keywords : A*ccelerometer mass, finite element, natural frequency, mode shape, thin plate.*

1.0 INTRODUCTION

Accelerometers are transducer that measures the acceleration of a vibrating body. It is widely used for vibration measurement because of its small size, high sensitivity and large usable frequency range. In most cases, systems that undergo experimental measurement are affected by transducer. This is true, especially if the test structure is small and lightweight. Then, the effect of accelerometer mass becomes significant when measuring on lightweight structure.

Lightweight structures are those structures that optimize the load carrying capacity of the elements by large deflection, allowing the load to be taken primarily in tension. It is characterized by having small mass relative to the

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applied load which the shape of the structure is determined through an optimization process. Lightweight structures include cable, membrane, shell, thin plate and folded structures.

Transducer effects on a structure are also known as 'mass loading' since the added mass applies an additional load to the structure. Døssing [1], in 1990s was the first to investigate this problem. It was found that the transducer apparent mass and its effects on the measurement values depends on the structure, the measuring location and frequency. His concerns were in determining the natural frequencies and dealing with the inconsistencies of the experimental data used for extracting mode shapes which was derived from the differences in the measuring location. Døssing introduced the driving point residue method to predict shifts of natural frequency due to mass loading effects.

Baldanzinni and Pierrini [2] have studied the effects of transducer mass and moments of inertia on frequency response function. They concluded that the transducer loading effects were mostly caused by transducer mass and not by moment of inertia. However, this conclusion only valid if low moment of inertia was applied to the structure. Mass loading is also very sensitive to ratio between local dynamic stiffness and transducer mass. A transducer placed on a nodal line did not caused mass loading effect, while it strongly influenced the measured data if place on anti nodal location.

2.0 THEORY OF MASS LOADING

The mass of an accelerometer can significantly affect the dynamic characteristics of the structure to which it is mounted. This is commonly called mass loading effect which tends to lower the measured natural frequencies. The general rules is the accelerometer mass should be less than one-tenth from the effective mass of the structure to which it is attached. Theoretically, the natural frequency is ;

$$\omega = \sqrt{\frac{k}{M}} \tag{1}$$

The addition of the accelerometer mass to the mass of the vibrating structure changes the resonant frequency of the vibrating systems as follows ;

$$f_m = f_s \sqrt{\frac{M}{M + m_a}} \tag{2}$$

where ω = natural frequency
k = stiffness of the structure
M = mass of the structure
m_a = accelerometer mass
f_m = frequency of the structure with the influence of the accelerometer mass
fs = frequency of the structure without the influence of the accelerometer mass

This relationship shows that if the accelerometer mass is kept small compared to the mass of the structure then any changes in the vibration will be only small.

The mass loading produced by accelerometer depends on the local dynamic properties of the structure [3]. The mass and resulting frequencies shift is proportional to the square of deflection of the associated mode. This study will determine how much the natural frequency will change due to the mass loading effect.

3.0 METHODOLOGY

The natural frequencies and mode shapes of the thin plate were obtained by using modal testing and finite element analysis.

3.1 Test structure

In this study, the aluminum alloy plate with 3mm thickness was used. The dimension of the test structures used in this study is 240mm x 230mm x 3mm and the mass of the plate is 447g. One edge clamped boundary condition was used to the test structure.

To investigate the mass loading effect of an accelerometer, the mass of the accelerometer was increased from 5.84 g to 45.84 g. The standard mass of multiple 10 gram were used in the experiment as added mass to the accelerometer mass. Here, we considered that the added mass is to be called the accelerometer mass to indicate the accelerometer mass effects. The accelerometer was attached at three different locations as shown in Figure 1. The accelerometer mass will be added until it was increased over than one-tenth of the mass of the plate. The accelerometer mass is identified as in Table 1.

Table 1 : Mass of the accelerometer mounted on the test plate

No	Mass of the accelerometer (g)	Percentage increase of the accelerometer mass compared to the plate mass (%)
1	5.84	1.3
2	15.84	3.5
3	25.84	5.8
4	35.84	8.0
5	45.84	10.2

3.2 Finite Element Analysis

ABAQUS Version 6.6.3 was used to obtain the natural frequency and its corresponding mode shape of the test structure. The test structure was modeled using shell element and while the accelerometer was modeled as hexagon solid element.. The model of accelerometer was attached on the test structure model at the locations as shown in Figure 1. The properties of the test structure are defined as follow;

Young's Modulus, $E = 72 \times 10^9$ Pa
Poisson Ratio, ν = 0.3
Density, ρ = 2700 kg/m^3

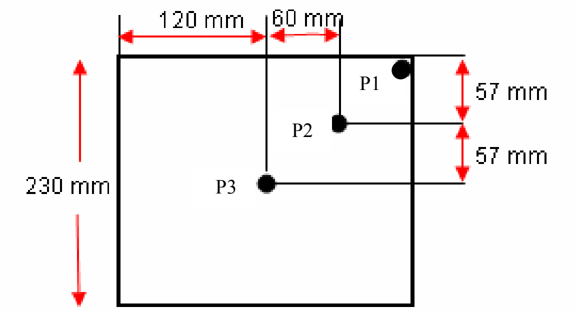


Figure 1 **:** Layout of accelerometer location on the plate

3.3 Experimental Modal Analysis

The accelerometer used for the experimental modal analysis is PiezoBeam accelerometer from KISTLER type 8636C50. It was chosen because of its small size and lightweight with a mass of 5.84 gram, thus the effect of mass loading is small and representing 1.3% of plate mass. An impact hammer method of excitation was used with KISTLER model 9722A500 impact hammer. It is equipped with low impedance force sensor to measure the force excited on the test structures. Plastic hammer tip was used to produce signal of the force pulse and to provide a broadband excitation to the interest frequency range of 0 – 2000 Hz. PAK MULLER-MK11 was used as the FFT analyzer to simultaneously measures the force and response, converted them into digital signals and computed their Discrete Fourier Transform. ME'scopeVES version 4.0 was used as modal analysis software to extract the natural frequencies and corresponding mode shapes from the measured data.

4.0 RESULTS OF THE NATURAL FREQUENCY

Tables 2 to 7 present the results obtained from numerical and experimental analysis for the first sixth natural frequency for tested plate.

4.1 Case 1 : Location of Accelerometer is at point 1 (P1)

Table 2 : Natural frequencies of plate from the numerical analysis

m_a (g)	% of m_a	Natural Frequencies (Hz) and Mode											
		1	% diff	2	% diff	3	% diff	4	% diff	5	% diff	6	% diff
0.0	0.0	45	0.0	113	0.0	275	0.0	372	0.0	407	0.0	723	0.0
5.84	1.3	44	2.2	108	4.4	272	1.1	348	6.5	398	2.2	698	3.5
15.84	3.5	42	6.7	102	9.7	266	3.3	321	13.7	394	3.2	665	8.0
25.84	5.8	41	8.9	97	14.2	260	5.5	308	17.2	392	3.7	647	10.5
35.84	8.0	39	13.3	93	17.7	254	7.6	301	19.1	391	3.9	637	11.9
45.84	10.2	38	15.6	91	19.5	250	9.1	297	20.2	391	3.9	629	13.0

Table 3 : Natural frequencies of plate from impact test

m_a (g)	% of m_a	Natural Frequencies (Hz) and Mode											
		1	% diff	2	% diff	3	% diff	4	% diff	5	% diff	6	% diff
5.84	1.3	43	0.0	102	0.0	254	0.0	340	0.0	401	0.0	664	0.0
15.84	3.5	42	1.6	96.8	5.1	253	0.4	316	7.1	397	1.0	641	3.5
25.84	5.8	41	4.5	92.6	9.2	251	1.2	303	10.9	387	3.5	631	5.0
35.84	8.0	40	6.6	89	12.7	248	2.4	293	13.8	376	6.2	630	5.1
45.84	10.2	39	8.5	86.3	15.4	244	3.9	287	15.6	376	6.2	-	-

4.2 Case 2 : Location of the accelerometer is at point 2 (P2)

Table 4 :Natural frequencies of plate from numerical analysis

m_a (g)	% of m_a	Natural Frequencies (Hz) and Mode											
		1	% diff	2	% diff	3	% diff	4	% diff	5	% diff	6	% diff
0.0	0.0	45	0.0	113	0.0	275	0.0	375	0.0	407	0.0	723	0.0
5.84	1.3	45	0.0	113	0.0	275	0.0	375	0.0	407	0.0	732	1.2
15.84	3.5	44	2.2	111	1.8	275	0.0	374	0.5	407	0.0	729	0.8
25.84	5.8	44	2.2	110	2.7	274	0.4	374	0.5	407	0.0	726	0.4
35.84	8.0	44	2.2	109	3.5	274	0.4	373	0.3	407	0.0	721	0.3
45.84	10.2	44	2.2	108	4.4	274	0.4	372	0.0	407	0.0	715	1.1

Table 5 : Natural frequencies of plate from impact test

m_a (g)	% of m_a	Natural Frequencies (Hz) and Mode											
		1	% diff	2	% diff	3	% diff	4	% diff	5	% diff	6	% diff
5.84	1.3	43	0.0	113	0.0	286	0.0	371	0.0	421	0.0	729	0.0
15.84	3.5	43	0.0	112	0.9	286	0.0	371	0.0	421	0.0	727	0.3
25.84	5.8	43	0.5	110	2.7	286	0.0	369	0.5	421	0.0	723	0.8
35.84	8.0	42	2.1	109	3.5	285	0.3	367	1.1	421	0.0	715	1.9
45.84	10.2	42	3.0	108	4.4	284	0.7	359	3.2	421	0.0	717	1.6

4.3 Case 3 : Location of the accelerometer is at point 3 (P3)

Table 6 : Natural frequencies of plate from numerical analysis

m_a (g)	% of m_a	Natural Frequencies (Hz) and Mode											
		1	% diff	2	% diff	3	% diff	4	% diff	5	% diff	6	% diff
0.0	0.0	45	0.0	113	0.0	275	0.0	372	0.0	407	0.0	723	0.0
5.84	1.3	45	0.0	113	0.0	275	0.0	370	0.5	407	0.0	716	1.0
15.84	3.5	45	0.0	113	0.0	271	1.5	363	2.4	407	0.0	701	3.0
25.84	5.8	44	2.2	113	0.0	267	2.9	357	4.0	407	0.0	689	4.7
35.84	8	44	2.2	113	0.0	263	4.4	353	5.1	407	0.0	678	6.2
45.84	10.2	44	2.2	113	0.0	259	5.8	349	6.2	407	0.0	670	7.3

Table 7 : Natural frequencies of plate from impact test

m_a (g)	% of m_a	Natural Frequencies (Hz) and Mode											
		1	% diff	2	% diff	3	% diff	4	% diff	5	% diff	6	% diff
5.84	1.3	43.3	0.0	114	0.0	285	0.0	370	0.0	-	-	715	0.0
15.84	3.5	43.3	0.0	114	0.0	279	2.1	361	2.4	-	-	697	2.5
25.84	5.8	43.2	0.2	114	0.0	276	3.2	355	4.1	-	-	682	4.6
35.84	8.0	43.2	0.2	114	0.0	271	4.9	350	5.4	-	-	669	6.4
45.84	10.2	43.1	0.5	114	0.0	267	6.3	346	6.5	-	-	660	7.7

5.0 ANALYSIS OF DATA AND DISCUSSION

5.1 General Rule for Accelerometer Mass

It was stated that, the general rules for the mass of the accelerometer is, it should be less than one-tenth of the mass of the structure to which it is attached. From Ashory [4], if the natural frequency shifted by 5% from the exact value, it was considered not acceptable and must be corrected to cancel the mass loading effect of accelerometer. For case 3, when the accelerometer mass was located at P3, increasing the accelerometer mass by 8% of the mass of the plate, the natural frequency of mode 4 was shifted by 5%. On the contrary, the natural frequency for some modes remains unchanged although the accelerometer mass was increased over 10% of the mass of plate. Therefore, the general rule for the mass of the accelerometer is not completely reliable in every case.

5.2 Varying Accelerometer Mass with Accelerometer Fixed at Location

The effect of the accelerometer mass on the natural frequency and mode shapes of the thin plate were investigated by comparing the results from finite element and experimental analysis. Figure 2 shows the graph of natural frequency against accelerometer mass located at P1 for mode 2. The graph shows that the natural frequency is inversely proportional to the accelerometer mass. The results of the finite element analysis and experimental modal analysis produce a similar trend

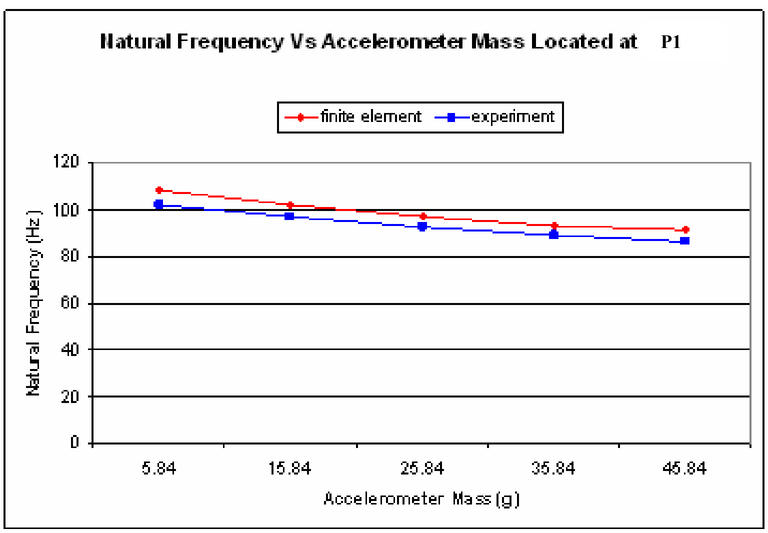


Figure 2 : Graph of natural frequency versus accelerometer mass located at P1 for mode 2

Figure 3 shows the second mode shapes obtained through numerical and experimental modal analysis. This figure clearly portrays the effect of accelerometer mass on thin plate. It shows that the changes to the mode shapes are more apparent for a larger accelerometer mass.

Thus, the effect of accelerometer mass on the natural frequency and its corresponding mode shapes shows clearly for the plates. The natural frequencies are proportional to the inverse of the accelerometer mass. The changed of the mode shapes are more obvious for a larger accelerometer mass.

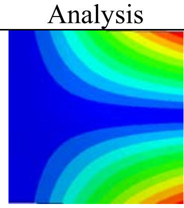
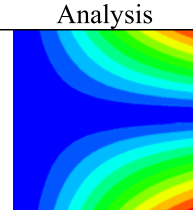
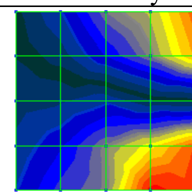
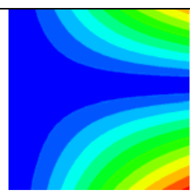
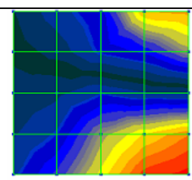
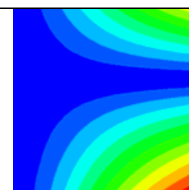
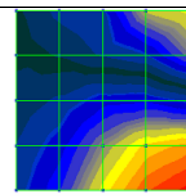
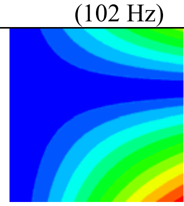
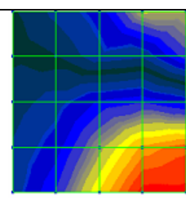
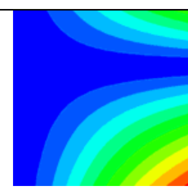
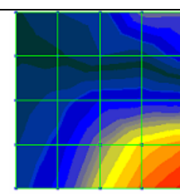
Numerical Analysis	Experimental Modal Analysis	Numerical Analysis	Experimental Modal Analysis
 (a) m_a : 0g (113 Hz)	NA	 (b) m_a : 5.84 g (108 Hz)	 (c) m_a : 5.84 g (102 Hz)
 (d) m_a : 15.85 g (102 Hz)	 (e) m_a : 15.84 g (96.8 Hz)	 (f) m_a : 25.84 g (97 Hz)	 (g) m_a : 25.84 g (92.6 Hz)
 (h) m_a : 35.84 g (93 Hz)	 (i) m_a : 35.84 g (89 Hz)	 (j) m_a : 45.84 g (91 Hz)	 (k) m_a : 45.84 g (86.3 Hz)

Figure 3 : Comparison of the mode shapes for mode 2, the accelerometer locate at P1.(NA-Not Aplicable; m_a - accelerometer mass)

5.3 Varying Accelerometer Location and Accelerometer Mass Constant

The effect of accelerometer location and accelerometer mass on thin plate was studied by comparing the natural frequency and its corresponding mode shapes of accelerometer mass; 5.84 gm and 45.84 gm. These mass was selected because 5.84 gm is the smallest mass, while 45.84 gm is the largest mass tested.

Table 8 shows the natural frequency of mode 2 from numerical and experimental analysis for the additional mass located at P1, P2 and P3.

Table 8: Comparison of natural frequencies of mode 2 for thin plate tested at different location of accelerometer.

Accelerometer Mass (m_a)	% of m_a	Natural Frequency (Hz)					
		Experiment			Finite Element		
		P5	P5	P13	P5	P9	P13
5.84 gm	1.3	102	113	114	108	113	113
45.84 gm	10.2	86.3	108	114	91	108	113

It is apparent that for different location of accelerometer mounted on the thin plate, different results were obtained. For 5.84 gm of accelerometer mass, the lowest natural frequency value is when accelerometer mounted at P1, while at P3 it gave highest value for the experimental results.

The table shows that for 45.84 gm of accelerometer mass at P2 and P3, the natural frequency shifted lower compared to 5.84 gm of accelerometer mass. While, for the accelerometer mounted at point P3 on thin plate, the natural frequency did not changed although the accelerometer mass was increased. This is due to the accelerometer being placed on a nodal line (zero deflection) for this corresponding mode as can be seen in Figure 3. The accelerometer located at P1, changed the natural frequency most compared to the other points because it is the maximum deformation for this mode. As discussed before, a transducer placed on a nodal line did not cause mass loading effect, while it strongly influenced the measured data if place at anti nodal location.

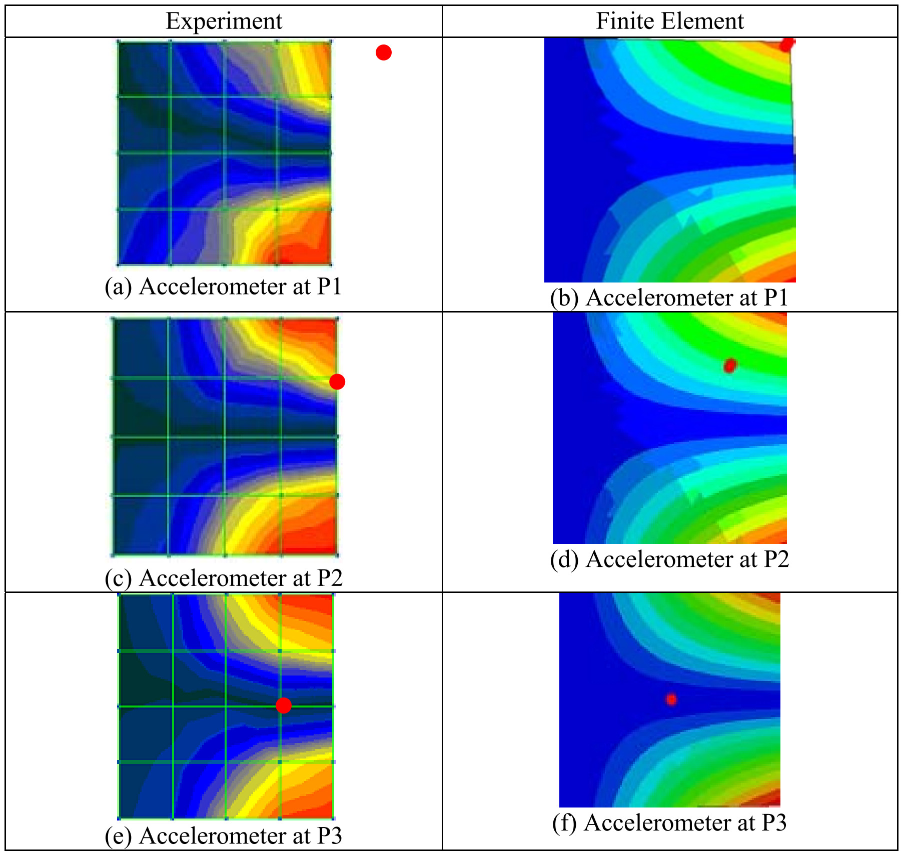


Figure 4 : Comparison of the mode shape for mode 2 with 5.84 gm of accelerometer mass located at P1, P2 and P3.

Figure 4 shows the mode shape of mode 2 for 5.84 gm of accelerometer mass obtained through experiment and finite element analysis. The effects on the mode shapes as the location of the accelerometer varied were portrayed clearly. The mode shapes for accelerometer located at P2 and P3 were looked similar. While, the mode shapes of the mode for accelerometer located at P1 was slightly different compared to the other location.

Figure 5 illustrates the mode shape of the mode 2 for thin plate with 45.84 gm of the accelerometer obtained through experiment and finite element analysis. The mode shapes of the mode were significantly different for different location of the accelerometer mounted on the thin plate although all the mode shapes were of the same mode.

It can be concluded that varying the location of accelerometer mounted on the thin plate, will give different results. An accelerometer placed on a nodal line did not cause mass loading effect, while it strongly influence the measured data if place at anti nodal location.

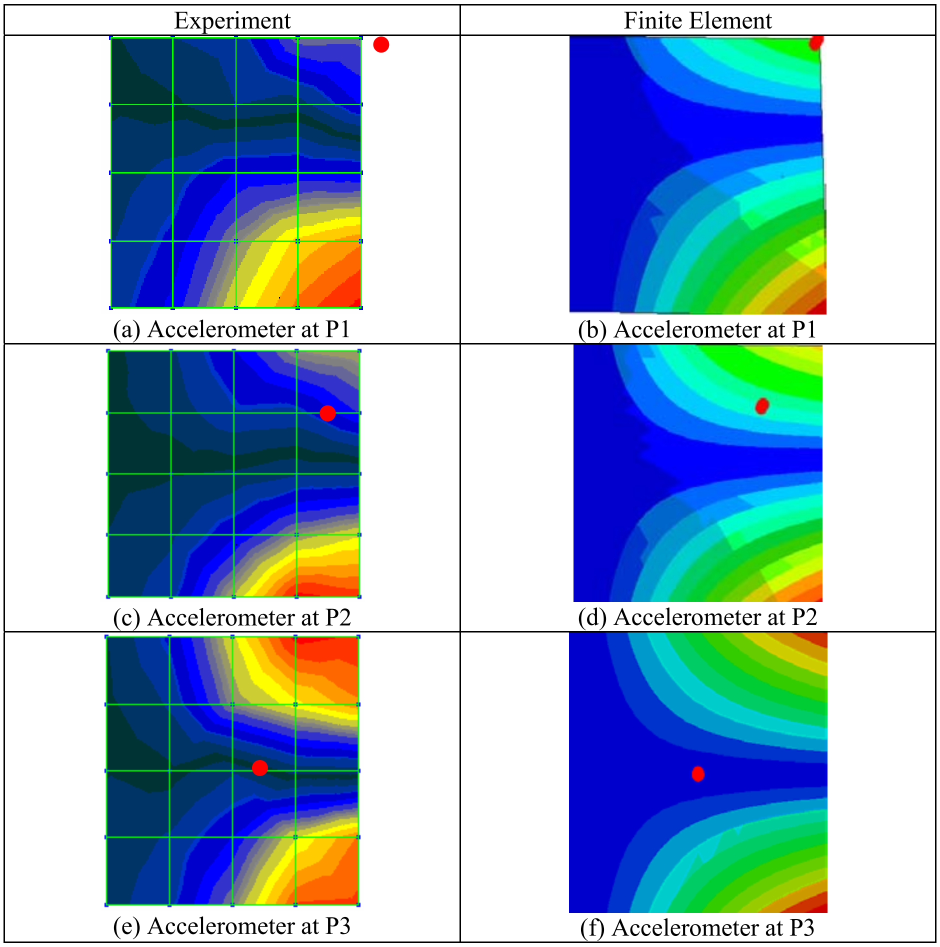


Figure 5 : Comparison of the mode shapes for mode 2 with 45.84g of the accelerometer mass located at P1, P2 and P3.

6.0 CONCLUSION

The objective of this study was to investigate the effect of accelerometer mass on thin plate vibration, where the natural frequencies and its corresponding mode shapes become the parameters of interest.

The results demonstrated that an accelerometer mass affect some of the modes of a structure while other modes remain unchanged. Accelerometer attached near the anti node has larger changes in natural frequency. Thus, the mode shapes for that particular mode also show significant changes. However for accelerometer attached near a nodal line of a particular mode of the structure, numerical results showed that the natural frequencies of the mode remain unchanged. As a result, no significant changes of mode shape at this particular mode.

It was noted that accelerometer mass should not be more than one-tenth of the mass of the structure [6]. This study showed that this rule may not be reliable for certain vibration mode. The vibration mode that was effect by mass loading must be corrected by using correction method presented in reference [7]. Thus, the lightest accelerometer mass as possible has to be used to decrease the mass loading effect.

It can be concluded that the effect of accelerometer mass depends on the location of the accelerometer, vibration mode and magnitude of the accelerometer mass. The studies on the effect of the accelerometer mass on the dynamic properties of the structures indicate that both natural frequency and mode shapes should be considered.

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